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THE PROBLEM OF INCREASING THE EFFICIENCY  
AND POWER OF GAS-TURBINE POWER PLANTS

By

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# UNEDITED ROUGH DRAFT TRANSLATION

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GAS-TURBINE POWER PLANTS

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English Pages: 15

SOURCE: Teploenergetika, No. 12, 1960, pp. 33-38.

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THE PROBLEM OF INCREASING THE EFFICIENCY  
AND POWER OF GAS-TURBINE POWER PLANTS\*

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Problems in improving gas-turbine power plants by decreasing the average temperature of the heat removed to the ambient medium are considered.

Two methods used at the present time for increasing the efficiency of gas turbine power plants can be cited: increasing the average temperature of the fuel heat fed to the working body or decreasing the average temperature of the heat removed to the ambient medium. While the average temperature of the heat input can be increased in regenerative cycles mainly by raising the upper temperature boundary of the cycle, the average temperature of the heat removal can be decreased only by additional use of the exhaust heat without lowering the lower temperature boundary of the cycle, since the latter boundary cannot be made lower than the

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\* Printed as a discussion. Editor.

temperature of the ambient medium.

An unpleasant feature of the gas-turbine cycle, distinguishing it from the steam-turbine cycle, is the fact that heat is obligatorily removed from the gases, when the temperature of the latter changes; this attests to the presence of great power losses accompanying the transfer of the exhaust heat to the ambient medium [1].

In Fig. 1 the line A-C conventionally depicts the process of heat removal from the gases, the area A-A'-C'-C-A is the quantity of heat removed along A-C, while the area A-D-C''-C-A is the power loss resulting from irreversibility of the heat removal. This loss can be written in the form [1].

$$\Pi_0 = i_A - i_c - T_0(s_A - s_c), \quad (1)$$

where  $T_0$  is the absolute temperature of the ambient medium.

Thus, for example, when  $T_0 = 290^\circ\text{K}$  and  $T_A = 500^\circ\text{K}$ , the power loss, calculated for 1 kg of combustion products, is equal to approximately 13 kcal/kg, which represents 20-40% of the effective work of the gas-turbine power plant. By inscribing the corresponding steam-power cycle (which we shall henceforth call the utilization cycle) into the contour A-D-C''-C-A, we can sharply decrease  $\Pi_0$  and produce an appreciable fuel economy.

The problem of using the exhaust heat of internal-combustion engines and gas turbines for power needs has a long history and up to now has not found any positive solution for two reasons. The first is the low thermodynamic efficiency of such utilization, while the second is the economically unjustified large heat-exchange surfaces of utilization boilers. Let us consider the first reason.

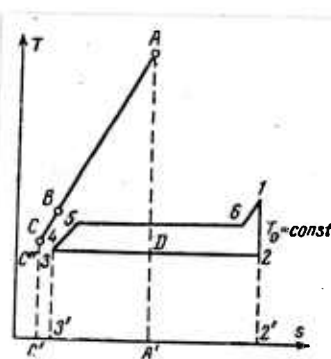


Fig. 1. Curves showing the heat exchange between the combustion products and water vapor.

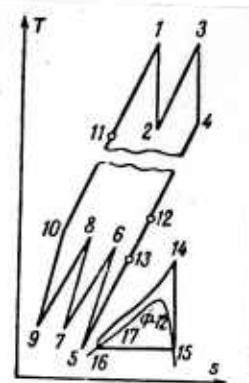


Fig. 2. Binary gas-freon cycle.

The low thermodynamic efficiency of steam utilization boilers is attributable to the presence of an isothermal heating section 5-6 (Fig. 1), which causes a large average temperature difference of the heat exchange between the exhaust gases and the water vapor. This difference can be decreased sharply by using in the utilization boiler supercritical-parameter steam, the heating curve of which has a configuration similar to the configuration of the cooling curve of the exhaust gases. However, it is not difficult to see that with the prevailing exhaust-gas temperatures of gas-turbine power plants water vapor with supercritical parameters cannot be used in a utilization boiler. Therefore, in our opinion, attention should be given to the use of low-boiling substances (such as freon, sulfur hexafluoride, perfluorobutane) in utilization boilers, substances that are sufficiently heat-resistant and have, when possible, a small critical pressure [2].

In order to obtain an idea of the expediency of using low-boiling substances as the working bodies of utilization power cycles comprising the lower stage of the binary gas-steam cycle, we made a number of calculations of a binary power plant, the upper boundary

of which is a GT-100-750 (IMZ) gas-turbine power plant, while the lower boundary is a power plant operating on freon F-12. For the appropriate calculations a p-i diagram drawn by G. Faltin for temperatures up to 800°C and pressures up to 120 atm abs [3] was used. The values of the internal relative efficiencies of all the turbines and the adiabatic efficiencies of the gas compressors were taken equal to 0.9; the adiabatic efficiency of the freon pump was 0.8; the mechanical efficiency of each unit was 0.98; the efficiency of the generator of the electric current for the gas-turbine unit was 0.99 and was 0.97 for the freon unit.

Figure 2 shows the binary cycle under consideration, while Fig. 3 shows the corresponding design for a binary power plant. Before entering the turbine the F-12 washes the surfaces of both intermediate coolers located between the compressor stages, as well as the surfaces of the additional freon heater, in which the gases leaving the regenerator are cooled from 187 to 120°C.

The efficiency of an autonomous GT-100-750 is 38%, while that of a binary power plant, whose freon-condensation temperature  $t_{15} = 20^\circ\text{C}$  ( $p_{15} = 5.78$  atm abs) and whose initial freon-stage parameters are 45 atm abs and 120°C, is 43.9%. In this case the combustion products leave the freon heater when  $t_{13} = 120^\circ\text{C}$ . A thermodynamic analysis by means of the entropy method shows that the freon adapter made it possible to reduce by 81% the power losses resulting from irreversible heat exchange between the combustion products and the ambient medium. The results of a comparison of an autonomous gas-turbine power plant and a binary gas-freon power plant (Fig. 3) are shown in the table.



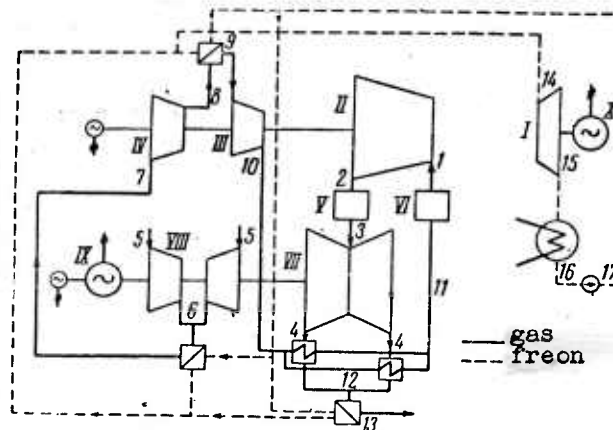


Fig. 3. Design of a binary gas-freon power plant.

I) Freon turbine ( $\epsilon = 7.5$ ;  $\eta_{oi} = 0.9$ ); II) high-pressure gas turbine ( $\epsilon = 2.45$ ;  $\eta_{oi} = 0.9$ ); III) high-pressure compressor ( $\epsilon = 2.52$ ;  $\eta_{ad} = 0.9$ ); IV) medium-pressure compressor ( $\epsilon = 2.52$ ;  $\eta_{ad} = 0.9$ ); V) low-pressure combustion chamber ( $\eta = 0.97$ ); VI) high-pressure combustion chamber ( $\eta = 0.97$ ); VII) low-pressure gas turbine ( $\epsilon = 7.15$ ;  $\eta_{oi} = 0.9$ ); VIII) low-pressure compressor ( $\epsilon = 3.16$ ,  $\eta_{ad} = 0.9$ ); IX) generator ( $N = 100$  Mw;  $\eta_g = 0.99$ ); X) generator ( $N = 15.408$  Mw;  $\eta_g = 0.97$ ). Parameters of the points:  
 1) 18.68 atm abs, 750°C; 2) 7.64 atm abs, 561°C;  
 3) 7.5 atm abs, 750°C; 4) 1.05 atm abs, 385°C;  
 5) 1.0 atm abs, 17.5°C; 6) 3.16 atm abs, 141°C;  
 7) 3.13 atm abs, 35°C; 8) 7.86 atm abs, 138°C;  
 9) 7.8 atm abs, 35°C; 10) 19.65 atm abs, 138°C;  
 11) point in front of the high-pressure combustion chamber; 12) 1.03 atm abs, 187°C; 13) 1.01 atm abs, 120°C; 14) 45 atm abs, 120°C; 15) 5.8 atm abs; 16) 5.78 atm abs, 20°C; 17) 47 atm abs, 25°C.

Inclusion of the freon stage reduced the power losses from heat exchange with the ambient medium by 9.36% by increasing the losses from heat exchange with freon by 1.91% (see table).

The idea of creating a cycle having all the heat-regeneration features inherent in the gas-turbine cycle but characterized by condensation of the working body, when the exhaust heat is removed from this body to the ambient medium, has been considered [3-5].

TABLE

Cause of Power Loss	Ratio of Power Loss to Fuel Heat, %				
	Gas por- tion, Fig. 3	Total design Fig.3	Design, Fig.4	Design, Fig.7	Design, Fig.8
Fuel combustion and irreversible heat exchange in the combustion chamber or in the boiler . .	37.35	37.35	42.06	42.55	39.91
Irreversible heat exchange between the combustion products and the freon or between the combustion products and water vapor. . . . .	-	1.91	-	-	3.16
Non-isentropic expansion in the turbines . . . .	4.37	5.43	2.29	5.58	5.47
Non-isentropic compression in the compressors and the turbines . . . . .	3.03	3.33	1.88	3.10	1.55
Irreversible heat exchange in the gas-gas regenerator. . . . .	2.19	2.19	17.38	5.16	-
Irreversible heat exchange between the exhaust working heat and the ambient medium. . . . .	11.86	2.52	2.25	2.08	1.69
Mechanical losses. . . . .	2.78	2.88	0.25	0.50	1.99
Losses in the electric-current generator . . . .	0.38	0.49	0.34	0.44	0.53
The sum of the power-loss factors, % . . . . .	61.96	56.10	66.45	56.41	54.30
Efficiency . . . . .	38.04	43.90	33.55	43.59	45.70

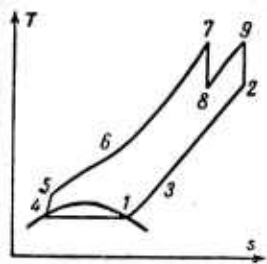


Fig. 4. Carbon-dioxide power cycle with ordinary regeneration.

Parameters of the points:

- 1) 60 atm abs, 21°C;
- 2) 66 atm abs, 589°C;
- 3) 62 atm abs, 61°C;
- 4) 60 atm abs, 21°C;
- 5) 240 atm abs, 47°C;
- 6) 225 atm abs, 441.5°C;
- 7) 210 atm abs, 650°C;
- 8) 118 atm abs, 569°C;
- 9) 107 atm abs, 650°C.

In this case, of course, the working body must be a low-boiling substance. Up to the present time the most stable low-boiling substance satisfying the requirements for the working bodies of power cycles has been carbon dioxide.

Figure 4 shows one of the previously proposed carbon-dioxide power cycles.

In this cycle the heating of the  $\text{CO}_2$  along the lines 6-7 and 8-9 takes place in a carbon-dioxide

boiler, which is also capable of operating on solid fuel. Upon leaving the low-pressure turbine, the  $\text{CO}_2$  is cooled along line 2-3 by heating the working body, after its emergence from the pump, along line 5-6. Process 3-4 takes place in a condenser.

Taking as the initial parameters of the cycle 650°C (at points 7 and 9),  $p_7 = 210$  atm abs,  $p_9 = 107$  atm abs,  $p_2 = 66$  atm abs,  $p_4 = 60$  atm abs ( $t_4 = 21^\circ\text{C}$ ) and taking the boiler efficiency to be equal to 0.925, the internal relative efficiency of the turbines to be 0.88, the adiabatic efficiency of the pump to be 0.75, the efficiency of the electric-current generator to be 0.99, and the mechanical efficiency to be 0.995, we obtain the electrical efficiency of the power plant  $\eta = 0.3355$ .

The table shows the characteristics of the heat balance of a carbon-dioxide power plant; these characteristics were compiled from the initial data chosen above. Comparing these characteristics

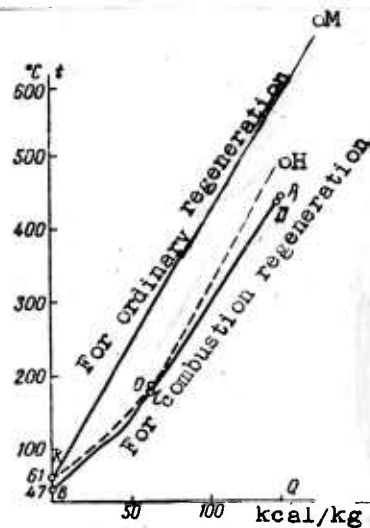


Fig. 5. Curves showing the heat exchange in the regenerator of a carbon-dioxide power plant.  
1) Ordinary regeneration;  
2) combination regeneration.

for the systems and cycles in Figs. 3 and 4, we can conclude that the power losses resulting from irreversibility of the heat exchange in the regenerator, in the case of a carbon-dioxide power plant, are 7.93 times greater than the analogous losses in a gas-freon power plant. The losses in a carbon-dioxide boiler are 1.13 times greater than the losses in the combustion chamber of a gas-

freon power plant. This is partially explained by the low initial temperature of the carbon-dioxide cycle ( $650^{\circ}\text{C}$  as opposed to  $750^{\circ}\text{C}$  in the gas-freon binary cycle). On the other hand, the losses from compression, in the case of a carbon-dioxide pump, were 1.77 times less than in a gas compressor or in the freon pump of a binary power plant.

The most appreciable difference between the power losses of individual power-plant units compared with each other occurs in the regenerator and is due to the relatively low degree of regeneration of the cycle (Fig. 4). Given the above-chosen initial parameters for this cycle, the degree of regeneration obtained is 0.76, when the temperature difference at the cold end of the regenerator is  $14^{\circ}\text{C}$ . At the same time, at the hot end of the regenerator a temperature difference of  $147.5^{\circ}\text{C}$  is established. The reason for this spread between the temperature differences at the ends of the regenerator is to be found in the fact that the heat capacity along

the heating line 5-6 is 75-100% greater than the heat capacity along the cooling line 2-3.

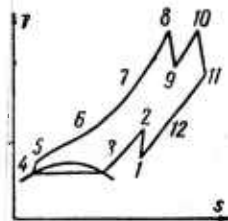


Fig. 6. Carbon-dioxide power cycle with combination regeneration.

Parameters of the points:

- 1) 17 atm abs, 164°C;
- 2) 63 atm abs, 184.2°C;
- 3) 62 atm abs, 61°C;
- 4) 60 atm abs, 21°C;
- 5) 240 atm abs, 47°C;
- 6) 236.5 atm abs, 176°C;
- 7) 225 atm abs, 435°C;
- 8) 210 atm abs, 650°C;
- 9) 72 atm abs, 514°C;
- 10) 66 atm abs, 650°C;
- 11) 18 atm abs, 489.6°C;
- 12) 17 atm abs, 164°C.

The dependence of the quantity of heat transferred to the regenerator on the change in the heat-exchange temperature difference is shown in Fig. 5. Here the heating line B-C-A (corresponds to line 5-6 in Fig. 4) in its lower part is so distorted relative to the cooling line M-K (corresponds to line 2-3 in Fig. 4) that a significant reduction of irreversibility by further rapprochement of the curves is impossible.

Dekhtyarev [6] spoke of the possibility of increasing the degree of regeneration in cycles of low-boiling substances by introducing combination heat regeneration, where a particular heating section of line 5-6 corresponds to two cooling sections, parallel to each other and transferring heat to the heating side of the regenerator. These two sections can be created by inserting at a specific point 1 of the cooling line (Fig. 6) an intermediate compression process 1-2 in an additional compressor, so that the heating of the working body along line 5-6 occurs as a result of simultaneous removal of heat on sections 2-3 and 12-1. Regenerative heating of the working body along line 6-7 will occur as a result of ordinary cooling (along line 11-12). The idea of combination regeneration thus consists in creating, by doubling the heat-re-

moval processes (on a particular temperature section), the effect of a corresponding increase in the heat capacity during heat removal.

In Fig. 5 the curve D-C-B represents the heating line in the regenerator of a power plant operating according to the cycle in Fig. 6 and possessing the same boundary parameters as the cycle in Fig. 4. The changes in the parameters of the cycle relate to points 11 and 1 ( $p_{11} = 18$  atm abs,  $p_1 = 17$  atm abs,  $t_1 = 64^\circ\text{C}$ ,  $p_2 = 63$  atm abs). The points D, C, and B in Fig. 5 correspond, respectively, to the points 7, 6, and 5 in Fig. 6. The cooling line H-O corresponds to line 11-12 in Fig. 6. Line O-K was obtained by adding the quantities of heat transferred by the  $\text{CO}_2$  on identical temperature segments of curves 12-1 and 2-3. This summation made possible a sharp rapprochement between the cooling section O-K and the heating section B-C (Fig. 5). As a result, the temperature difference at the hot end of the regenerator decreased sharply (from  $147.5$  to  $54.6^\circ\text{C}$ ), while at the same time there was a noticeable decrease in the average difference in the heat-exchange temperature in the regenerator, and the degree of regeneration rose up to  $0.87$ .

Figure 7 shows a design, the cycle of which is depicted in Fig. 6. The numbers on both diagrams correspond to each other. Given the above-chosen initial parameters, the electrical efficiency  $\eta$  of the power plant (Fig. 7) equals  $0.4359$ . The characteristics of the heat balance of the design in Fig. 7 are shown in the table.

A comparison of the heat balances shows that combination heat regeneration (Fig. 6) reduced the power losses resulting from irreversibility of regenerative heat exchange by a factor of almost  $3.5$  in comparison with ordinary regeneration (Fig. 4).

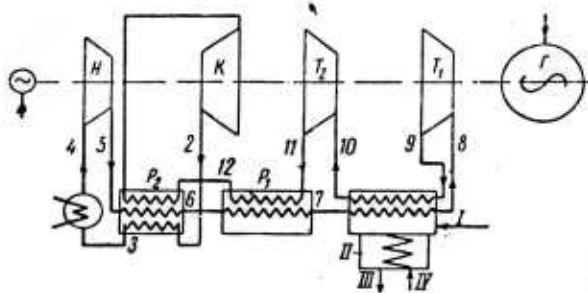


Fig. 7. Design of a carbon-dioxide power plant with combination regeneration (for parameters of the points see Fig. 6).  
I) Fuel; II) air heater;  
III) gases; IV) air.

This was the main reason for the corresponding fuel economy equal to 23%.

According to Dekhtyarev [6] the specific surfaces of the regenerators of a carbon-dioxide power plant do not exceed the limits assumed for the regenerators of closed gas-turbine plants.

They allow considerably lower average values for the temperature difference of regenerative heat exchange than is possible in the case of closed gas-turbine plants, owing to the considerably larger pressure values on both sides of the heat-exchange surfaces.

A carbon-dioxide power plant of simple design can have large unit capacities (of the order of 500 Mw or more) on each exhaust; this is due to the large specific-gravity values of carbon dioxide when under high pressure in the power plant. The great efficiency of the plant makes its use particularly expedient in areas involving solid fuel. It is of interest to study a carbon-dioxide power plant equipped with a pressure-charging boiler and making it possible to replace the steam-water portion of a steam-gas power plant with a carbon-dioxide portion; this should decrease the heating surfaces of the boiler by a factor of several units and should noticeably increase the efficiency of the power plant.

In conclusion, let us consider the case of utilization of the exhaust gases of a gas-turbine power plant to heat the feed water in a steam-water power plant with super-critical initial parameters.

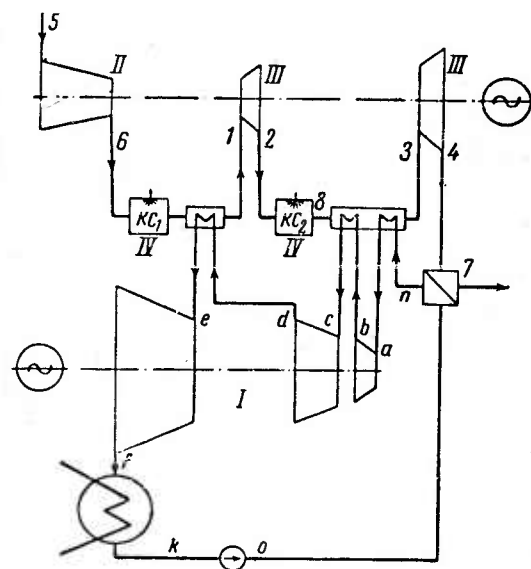


Fig. 8. OTI Juza steam-gas design using supercritical parameter water vapor.  
 1) Steam turbine; II) compressor; III) gas turbine; IV) combustion chamber. Parameters of the points: 1) 7.68 atm abs, 750°C; 2) 2.88 atm abs, 546°C; 3) 2.8 atm abs, 750°C; 4) 1.05 atm abs, 546°C; 5) 1 atm abs, 17 °C; 6) 8.07 atm abs, 276°C; 7) 98°C; a) 300 atm abs, 650°C; b) 55 atm abs; c) 49.5 atm abs, 565 °C; d) 15 atm abs; e) 13.95 atm abs, 565°C; o) 380 atm abs, 26.7°C; n) 400°C.

Figure 8 shows one of the variants of a design for a corresponding steam-gas power plant made in the course of developing Juza's idea [7]. For an approximate calculation a combination variant of a GT-100-750 unit with an SKK-300 unit was chosen. An important distinction between the design in Fig. 8 and ordinary designs involving high-pressure steam generators [8] is that in the former water vapor with supercritical initial parameters is used. According to the design in Fig. 8 superheating of the water vapor along lines n-a and b-c takes place in combustion chamber  $KC_2$  or immediately after it, while superheating of this steam along line d-e takes place in



combustion chamber  $KC_1$ . An approximate calculation of the power plant under study (the results of which are given in the table and in Fig. 9) was made under the condition that all the heating of the water vapor after the boiler-utilizer takes place only in  $KC_2$ .

In the calculations the efficiency of the combustion chamber was taken to be 0.97, the mechanical efficiency 0.98, the efficiency of the electric-current generator 0.99, the adiabatic efficiency of the pump 0.8. The multiplicity factor of the water vapor in relation to the combustion products was equal to 0.2293 kg/kg; the electrical efficiency of the steam-gas power plant was found to be 45.7%, which is 6.8% higher than the value of this efficiency for an autonomous SKK-300 (with nine regenerative heaters and a boiler efficiency of 0.9) and 16.7% higher than the electrical efficiency of an autonomous GT-100-750 (LMZ). The power of the steam-turbine portion of a combination power plant operating according to the design in Fig. 8 is 2.29 times greater than the power of its gas-turbine portion. Thus adding a gas-turbine portion to the SKK-300 according to the diagram in Fig. 8 will increase the power of the steam-gas power plant up to 430 Mw.

Figure 9 shows curves illustrating the relation between the quantities of heat transferred and the changes in the temperatures of the combustion products and the water vapor during their heat exchange. Rough calculations showed that in the case of a power plant operating according to the design in Fig. 8 the specific heating surface (including the heating surfaces of the combustion-chamber shields and the boiler-utilizer) is approximately  $0.071 \text{ m}^2/\text{kW}$ .

The reasons for the different efficiencies of the designs considered above are readily understood from an examination of the

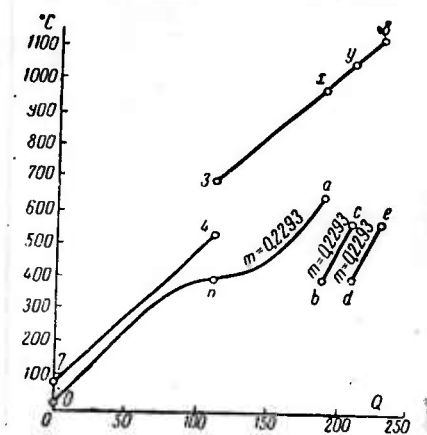


Fig. 9. Curves showing the heat exchange between gas and vapor in the OTI-Juza arrangement.

data in the table, which contains a summary of all the power losses occurring in the designs being studied. Of all the designs considered gas-turbine power plants have minimum power losses accompanying the burning of the fuel and the transfer of its heat to the working body; these losses are 2.6-4.8% higher in the case

of other power plants. However, this increase in the case of the other power plants is more than compensated by a decrease in the power losses accompanying the removal of exhaust heat to the surrounding medium; this decrease amounts to 9.34-10.17%. Now it is clear why the designs in Figs. 3, 7, and 8 are much more economical than the autonomous GT-100-750.

The data in the table indicate the importance of the problem of decreasing the irreversibility of the heat exchange between the working body and the ambient medium.

### Conclusions

1. A significant decrease in the irreversibility of the heat exchange between the exhaust gases of a gas-turbine power plant and the ambient medium can be achieved by creating combination steam-gas power plants, in which the steam portion operates with supercritical initial parameters. By using a freon cycle as the utilization steam power cycle the relative fuel economy can amount to 10-13%. The economy turns out to be even greater, when a gas-turbine

power plant is combined with steam-water power plants of type SKK.

2. The use of combination heat regeneration in carbon-dioxide power plants makes them very promising for medium and high capacities. Of special interest is the possibility of obtaining several hundred megawatts per exhaust.

3. The examples given in our article are not optimum ones. In order to achieve the maximum economic effect in the designs considered, it is necessary to carry out theoretical and experimental investigations, the purpose of which is to ascertain the optimum parameters of the designs and to obtain the optimum technological and economic indices.

#### REFERENCES

1. GOKHSHEYN, D. P. Entropiynyy metod rascheta energeticheskikh poter' (The Entropy Method of Calculating Power Losses). Gosenergoizdat (State Power-engineering Publishing House), 1951.

2. GOKHSHEYN, D. P. "Atomnaya energiya" (Atomic Energy). Supplement to No. 1, 1958.

3. FALTIN, G. Energietechnik, No. 10, 1953.

4. GOKHSHEYN, D. P. Patent No. 59491, Class 14H, 6, 1941.

5. GOKHSHEYN, D. P. Article in a Collection of Works of the Commission on Technical Thermodynamics of the Institute of Power Engineering of the Academy of Sciences of the USSR, 1940.

6. DEKHTYAREV, V. L. Theses of Reports at the Enlarged Session of the Commission of the State Scientific and Technical Committee of the Council of Ministers of the Ukr. SSR on the Introduction of Gas Turbines into the Power Economy of the Ukraine. Publishing House of the State Scientific and Technical Committee of the Ukr. SSR, 1959.

7. JUZA. V<sup>1</sup>yvoi Spalovacich Turbin, Strojinstvi, BSR, 147, 1955.

8. LOZHKIN, A. N. and GEL'TMAN, A. E. "Kotloturbostroeniye" (Boiler and Turbine Construction), No. 1, 1949.